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INTERNAL GEAR PUMP

Cross-Reference to Prior Application

This is a U.S. National Phase Application under 35 U.S.C. §371 of International Patent Application No. PCT/JP2004/019253, filed December 22, 2004, and claims the benefit of Japanese Patent Application No. 2003-435370, filed December 26, 2003, both of which are incorporated by reference herein. The International Application was published in Japanese on July 14, 2005 as International Publication No. WO 2005/064163 under PCT Article 21(2).

Technical Field

The present invention relates to an internal gear pump which draws and discharges fluid by changes in the volumes of cells formed between an inner rotor and an outer rotor when the inner rotor and the outer rotor rotate in engagement with each other.

Background Art

Conventionally, an internal gear pump comprises an inner rotor having "n" external teeth, an outer rotor formed

with "n+1" internal teeth which are engageable with the external teeth, and a casing in which a suction port for drawing fluid and a discharge port for discharging fluid are formed. Also, in a state in which the inner rotor and the outer rotor are accommodated in a hole formed in the casing, the inner rotor connected to a driving shaft is rotated, thereby rotating the outer rotor while the external teeth engage the internal teeth, so that fluid is drawn and discharged by changes in the volumes of a plurality of cells formed between the inner rotor and the outer rotor.

The cells are separately delimited at a front portion and at a rear portion as viewed in the direction of rotation of the inner and outer rotors by the external teeth of the inner rotor and the internal teeth of the outer rotor which contact each other, thereby forming independent fluid conveying chambers which is rotatably moved as the inner rotor rotates. Each cell becomes the smallest in volume in the course of engagement between the external teeth and the internal teeth, and the cell is then increased in volume as it moves along the suction port, thereby drawing fluid through the suction port. Also, the cell having the largest volume is decreased in volume as it moves along the discharge port, thereby discharging fluid through the discharge port (refer to Japanese Patent No. 3293507).

Also, the inner rotor and the outer rotor are disposed to have a predetermined eccentric distance therebetween, and the outer rotor is disposed coaxially with the hole of the casing.

Since such internal gear pumps having the above construction are compact and simply constructed, it is widely used as pumps for lubricating oil in automobiles and as oil pumps for automatic transmissions, etc. When an internal gear pump is mounted on an automobile, a means for driving the internal gear pump includes a crankshaft directly-connected and driving method in which an inner rotor directly connected to a crankshaft of an engine is driven by the rotation of the engine.

Meanwhile, it is common that the respective members of such an internal gear pump are dimensioned to have predetermined play when the gear pump is manufactured for the purpose of convenience of assembling or the like. Specifically, the internal diameter of a through-hole bored in a central portion of the inner rotor is set to be about 0.1 mm to 0.06 mm larger than the external diameter of the driving shaft loaded into the through-hole, and the internal diameter of the hole formed in the casing is set to be about

0.1 mm to 0.6 mm larger than the external diameter of the outer rotor. Also, as previously mentioned, the outer rotor and the hole of the casing are disposed coaxially with each other, and a clearance of about 0.05 mm to 0. 3 mm is set between an entire outer peripheral surface of the outer rotor and an inner peripheral surface of the hole formed in the casing.

Accordingly, when the internal gear pump constructed as above is driven, the inner rotor is rotated while being whirled by about 0.05 mm to 0.30 mm in its radial direction from an central axis of the driving shaft, and due to the clearance between the inner rotor and the driving shaft, the outer rotor is rotated while being whirled by about 0.05 mm to 0.3 mm in its radial direction from an central axis of the hole of the casing. For this reason, the external teeth of the inner rotor may collide against the internal teeth of the outer rotor, and the driving force received by the outer rotor at the time of this colliding may cause the outer peripheral surface of the outer rotor to further collide against the inner peripheral surface of the hole formed in the casing. Accordingly, noise may be generated when the internal gear pump is driven, and the pump efficiency may be deteriorated.

As a means for suppressing the occurrence of such collision between the outer peripheral surface of the outer rotor and the inner peripheral surface of the hole formed in the casing, there has conventionally been employed a construction in which a faucet is formed at a radial central portion of the inner rotor, and the faucet is loaded into a groove formed at the bottom of the hole of the casing to suppress the whirling of the inner rotor, thereby avoiding the occurrence of the collision between the outer peripheral surface of the outer rotor and the inner peripheral surface of the hole of the casing.

[Patent Document 1] Japanese Patent No. 3293507

Summary~~Disclosure~~ of the Invention

However, there has conventionally been a limit to higher efficiency of the gear pump because sliding resistance is generated between the faucet formed in the inner rotor and the groove formed in the casing, an energy loss caused by the sliding resistance occupies about 25% of the total energy loss that is caused when the internal gear pump is driven.

The present invention has been made in view of the

above circumstances. Accordingly, it is an object of the present invention to provide an internal gear pump in which the sliding resistance of the internal gear pump can be reduced, and the occurrence of noise and a decrease in the pump efficiency can be suppressed to the minimum even in this construction.

In order to solve the above problems and achieve the above object, the present invention proposes the following; ~~means.~~

~~Means for Solving the Problems~~

That is, the present invention provides an internal gear pump having~~comprising~~: an inner rotor formed with "n" external teeth ("n" is a natural number); and an outer rotor formed with (n+1) internal teeth which are engageable with the external teeth, and a casing formed with a suction port for drawing fluid and a discharge port for discharging fluid. The fluid is conveyed by drawing and discharging the fluid by changes in volumes of cells formed between tooth surfaces of the inner rotor and the outer rotor while the inner rotor and the outer rotor rotate in engagement with each other. The internal diameter of a hole formed in the casing for accommodating the inner rotor and the outer rotor is set to

be 0.1 mm to 0.6 mm larger than that the external diameter of the outer rotor. When "er" is an eccentric distance between the inner rotor and the outer rotor and "eh" is an eccentric distance between the inner rotor and the hole formed in the casing, the following inequality is satisfied:

$$0.005 \text{ mm} \leq (eh - er) \leq 0.030 \text{ mm}$$

According to the present invention, the distance between the inner peripheral surface of the hole formed in the casing and the outer peripheral surface of the outer rotor at an engaging position where the volume of a cell which is defined when the external teeth of the inner rotor the internal teeth of the outer rotor rotate in engagement with each other is the smallest is suppressed to the minimum.

Accordingly, when the driving force is transmitted from the external teeth of the inner rotor to the internal teeth of the outer rotor, and the outer rotor is moved forward in the direction of rotation thereof in a direction tangential to the engaging position of the outer rotor, and rotated along the inner peripheral surface of the hole of the casing, the forward movement of the outer rotor in the direction of rotation thereof is restrained by the inner peripheral surface of the hole of the casing.

As a result, because the position of the outer rotor disposed inside the hole of the casing becomes stable, the occurrence of collision between the external teeth of the inner rotor and the internal teeth of the outer rotor, and the occurrence of collision between the outer peripheral surface of outer rotor and the inner peripheral surface of the hole of the casing can be suppressed to the minimum. Further, even if such collisions occur, the collision energy at that time can be suppressed to the minimum.

Further, the outer rotor is restrained in the forward movement in the direction of rotation thereof, and moved along the inner peripheral surface of the hole of the casing, so that the distance of movement of the outer rotor toward the position opposite to the engaging position with respect to the rotation of center becomes small by the restrained distance of the outer rotor. As a result, the occurrence of collision between the external teeth of the inner rotor and the internal teeth of the outer rotor at the opposite position can be suppressed.

Moreover, since the distance between the inner peripheral surface of the hole of the casing and the outer peripheral surface of the outer rotor at the opposite position can be ensured to the maximum, the occurrence of

collision between the outer peripheral surface of the outer rotor and the inner peripheral surface of the hole of the casing at the opposite position can be suppressed.

From the above description, even if the construction in which a faucet is formed in the radial central portion of the inner rotor and the faucet is loaded into a groove formed at the bottom of the hole of the casing is employed, the occurrence of collision between the outer rotor and the casing and between the external teeth of the inner rotor and the internal teeth of the outer rotor can be suppressed. Accordingly, the sliding resistance of the internal gear pump can be remarkably reduced, and the occurrence of noise and a decrease in the pump efficiency can be suppressed to the minimum even in this construction.

According to the internal gear pump of the present invention, the sliding resistance of the internal gear pump can be remarkably reduced, and the occurrence of noise and a decrease in the pump efficiency can be suppressed to the minimum even in this construction.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a plan view showing an internal gear pump

according to an embodiment of the present invention;

~~FIG. 2~~ FIGS. 2A and 2B are tables ~~is a table~~ including data values shown as Example 1 of the internal gear pump shown in FIG. 1; and

FIG. 3 is a table including data values according Example 2 of the present invention.

Detailed Description of~~Best Mode for Carrying out~~ the Invention

An internal gear pump according to an embodiment of the present invention will now be described with reference to FIGS. 1 and 2.

The internal gear pump shown in FIG. 1 generally comprises an inner rotor 10 formed with eight external teeth 11, an outer rotor 20 formed with nine internal teeth 21 which are engageable with the external teeth 11, and a casing 30 in which a suction port for drawing fluid and a discharge port for discharging fluid are formed.

Also, the casing 30 is formed with a hole 31 which accommodates the inner rotor 10 and the outer rotor 20. Here, the illustration of the suction port and the discharge port are omitted in FIG. 1.

A through-hole 12 is bored in a radial central portion of the inner rotor 10. The internal diameter of the inner rotor 10 is set to be about 0.1 mm to 0. 6 mm larger than the external diameter of a driving shaft loaded into the through-hole 12. In addition, the driving shaft is directly connected to a crankshaft of an engine (not shown), so that the inner rotor 10 is supported so as to be rotatable about an axis O1 in the peripheral direction inside the casing 30 by the rotation of the engine. Accordingly, the axis O1 is the center of rotation of the through-hole 12 as well as the centers of rotation of the inner rotor 10 and the driving shaft.

The outer rotor 20 is disposed so that the internal teeth 21 are engageable with the external teeth 11 in a state that an axis O2 of the outer rotor is offset (eccentric distance er) from the axis O1 of the inner rotor 10, and is supported so as to be rotatable about the axis O2 in the peripheral direction inside the casing 30.

Here, the external teeth 11 of the inner rotor 10 are formed so that the tooth profiles of tooth tips 11a thereof are formed based on an epicycloid curve which is generated by rolling a first circumscribed-rolling circle Ai along a

first base circle Di without slip, and the tooth profiles of tooth spaces 11b thereof are formed based on an hypocycloid curve which is generated by rolling a first inscribed-rolling circle Bi along the first base circle Di without slip.

Further, the internal teeth 21 of the outer rotor 20 are formed so that the tooth profiles of tooth tips 21a thereof are formed based on a hypocycloid curve which is generated by rolling a second inscribed-rolling circle Bo along a second base circle Do without slip, and the tooth profiles of tooth spaces 21b thereof are formed based on an epicycloid curve which is generated by rolling a second circumscribed-rolling circle Ao along the second base circle Do without slip.

Meanwhile, the eccentric distance er between the axis O1 of the inner rotor 10 and the axis O2 of the outer rotor 20 is obtained such that the following formula is satisfied:

$$er = (d-D)/4$$

where 'd' is the diameter of a circle obtained by sequentially connecting apexes of the tooth tips 11a of the respective external teeth 11 of the inner rotor 10 with each other, i.e., the larger diameter of the inner rotor, and 'D'

is the diameter of a circle obtained by sequentially connecting bottoms of the tooth spaces 11b of the respective external teeth 11 of the inner rotor 10 with each other, i.e., the smaller diameter of the inner rotor 10.

The inner rotor 10 and the outer rotor 20 are rotated by the rotation of the driving shaft while the tooth surfaces thereof engage each other. Cells S serving as fluid conveying chambers are formed between engaging points of the inner rotor 10 and the outer rotor 20 which are engageable with each other. The suction port and the discharge port opened to the cells S are formed in the casing 30, so that the exchange of fluid with the respective cells S through the suction port and the discharge port is performed.

As the inner rotor 10 and the outer rotor 20 rotate, the cells S are also rotatably moved to cause changes in the volumes of the cells, so that fluid is drawn through the suction port in the course of an increase in the volumes of the cells S, while fluid is discharged through the discharge port in the course of a decrease in the volumes of the cells S.

The casing 30 is formed with the hole 31 which

accommodates the inner rotor 10 and the outer rotor 20, as previously mentioned. The internal diameter of the hole 31 is set to be about 0.1 mm to 0.6 mm larger than the external diameter of the outer rotor 20. Also, the center O3 of the hole 31, as shown in FIG. 1, is located so that it is offset from the axis O2 of the outer rotor 20 by 0.005 mm to 0.030 mm, more preferably, 0.010 mm to 0.020 mm in a direction away from the axis O1 of the inner rotor 10 and the engaging position A with respect to the axis O2. In other words, when 'er' is the eccentric distance between the inner rotor 10 and the outer rotor 20, and 'eh' is the eccentric distance between the inner rotor 10 and the hole 31 of the casing 30, the following formula is satisfied:

$$0.005 \text{ mm} \leq (eh - er) \leq 0.030 \text{ mm}$$

In addition, as shown in FIG. 1, the engaging position A means a position where the rotational driving force from the inner rotor 10 is transmitted to the outer rotor 20.

As a result, as shown in FIG. 1, a clearance t between an inner peripheral surface of the hole 31 of the casing 30 and an outer peripheral surface of the outer rotor 20 is gradually decreased from the engaging position A toward a peripheral position B offset by 180° forward or rearward in the direction of rotation from the position A so that a clearance tA is the largest at the engaging position A and a

clearance tB is the smallest at the peripheral position B.

From the above description, the clearance tA at the engaging position A is set to 0.020 mm to 0.295 mm, and the clearance tB at the peripheral position B is set to 0.055 mm to 0.330 mm.

In addition, in FIG. 1, the hole 31 of the casing 30 is shown in an enlarged state for the purpose of convenience of explanation.

Example 1 including data values related to the inner rotor 10, the outer rotor 20 and the casing 30, which are shown in FIG. 1, are shown in FIGS. 2A and 2B along with Comparative example 1 as a related art.

In FIG. 2A, the larger diameter of the outer rotor means the diameter of a circle obtained by sequentially connecting the bottoms of the tooth spaces 21b of the respective internal teeth 21 with each other in the peripheral direction, and the smaller diameter of the outer rotor means the diameter of a circle obtained by sequentially connecting the apexes of the tooth tips 21a of the respective internal teeth 21 with each other in the peripheral direction.

In both Example 1 and Comparative example 1, it was noted herein that the internal diameter of the hole formed in the casing for accommodating the inner rotor and the outer rotor was set to 79.99 mm to 80.01 mm, and the external diameter of the outer peripheral surface of the outer rotor opposing the inner peripheral surface of the hole was set to 79.75 mm to 79.80 mm.

As shown in FIG. 2B, the positions of the inner rotor 10 and the hole 31 (or axis O3) of the casing 30 in Example 1 were not varied as compared with the related art. Thus, only the outer rotor 20 (axis O2) was offset by 0.015 mm toward the engaging position A, i.e., downward in FIG. 1 so that the eccentric distance er was 0.015 mm smaller than that in Comparative example 1. Also, a gap was generated at the engaging position A between the tooth tips 11a of the external teeth 11 and the tooth spaces 21b of the internal teeth 21 by the offset distance, and the gap between the tooth tips 11a of the external teeth 11 and the tooth spaces 21b of the internal teeth 21 was decreased at the peripheral position B, which resulted in a change in an engaging state of the gear pump. Thus, in order to maintain the engaging state, as shown in FIG. 2A, the diameter of the first circumscribed-rolling circle Ai which generated the tooth

tips 11a of the external teeth 11 was made small by 0.030 mm, and the diameter of the second circumscribed-rolling circle Ao which generated the tooth spaces 21b of the internal teeth 21 was made small by 0.030 mm.

As previously mentioned, the eccentric distance er was calculated from the larger diameter d and the smaller diameter D of the inner rotor, that was, the tooth heights of the inner and outer rotors was 0.015 mm smaller as that in Comparative example 1 as a related art, whereby Example 1 could be realized.

From the above description, the clearance tA between the inner peripheral surface of the hole 31 of the casing 30 and the outer peripheral surface of the outer rotor 20 at the engaging position A could be suppressed to the minimum, and the engaging state between the external teeth 11 and the internal teeth 21 could be maintained even in this construction similarly to the related art.

As described above, according to the internal gear pump of the present embodiment, the clearance tA between the inner peripheral surface of the hole 31 formed in the casing 30 and the outer peripheral surface of the outer rotor 20 could be suppressed to the minimum at the engaging position

A where the external teeth 11 of the inner rotor 10 and the internal teeth 21 of the outer rotor 20 engaged each other to form the cell S having the smallest volume.

Accordingly, when the driving force was transmitted from the external teeth 11 of the inner rotor 10 to the internal teeth 21 of the outer rotor 20, and the outer rotor 20 was moved forward in the direction of rotation thereof in a direction tangential to the engaging position A of the outer rotor 20, and moved along the inner peripheral surface of the hole 31 of the casing 30, the forward movement of the outer rotor in the direction of rotation thereof was restrained by the inner peripheral surface of the hole 31 of the casing 30.

As a result, since the position of the outer rotor 20 disposed inside the hole 31 of the casing 31 became stable, the occurrence of collision between the external teeth 11 of the inner rotor 10 and the internal teeth 21 of the outer rotor 20 and the occurrence of collision between the outer peripheral surface of the outer rotor 20 and the inner peripheral surface of the hole 31 of the casing 30 could be suppressed to the minimum. Even if such collisions occur, the collision energy at that time could be suppressed to the minimum.

Further, the outer rotor 20 was restrained in the forward movement in the direction of rotation thereof, and moved along the inner peripheral surface of the hole 31 of the casing 30, so that the distance of movement of the outer rotor 20 toward the peripheral position B offset by 180° from the engaging position A forward or rearward in the direction of rotation thereof became small by the restrained distance of the outer rotor, that was, the outer rotor 20 was biased to the peripheral position B. As a result, the occurrence of collision between the external teeth 11 of the inner rotor 10 and the internal teeth 21 of the outer rotor 20 at the peripheral position B could be suppressed to the minimum.

Moreover, since the distance tB between the inner peripheral surface of the hole 31 of the casing 30 and the outer peripheral surface of the outer rotor 20 at the peripheral position B could be ensured to the maximum, the occurrence of collision between the outer peripheral surface of the outer rotor 20 and the inner peripheral surface of the hole 31 of the casing 30 at the peripheral position B could be suppressed.

From the above description, even if the construction in

which a faucet was formed in the radial central portion of the inner rotor 10 and the faucet was loaded into a groove formed at the bottom of the hole 31 of the casing 30 was employed, the occurrence of collision between the outer rotor 20 and the casing 30 and between the external teeth 11 of the inner rotor 10 and the internal teeth 21 of the outer rotor 20 could be suppressed. Accordingly, the sliding resistance of the internal gear pump could be remarkably reduced, and the occurrence of noise and a decrease in the pump efficiency could be suppressed to the minimum even in this construction.

The technical scope of the present invention is not limited to the aforementioned embodiment, but various modifications can be made without departing from the spirit of the present invention.

In the above embodiment, as the tooth profiles of the external teeth 11 of the inner rotor 10 and those of the internal teeth 21 of the outer rotor 20, the tooth profiles formed based on the cycloid curves are exemplified. However, the present invention is not limited thereto, and may adopt tooth profiles formed based on, for example, trochoid curves. Example 2 including data values according to the present invention in this case is shown in FIG. 3, along with

Comparative example 2 as a related art. In Both Example 2 and Comparative example 2 shown in FIG. 3, the internal diameter of a hole formed in a casing for accommodating inner and outer rotors is set to 59.99 mm to 60.01 mm, the external diameter of an outer peripheral surface of the outer rotor opposing an inner peripheral surface of the hole is set to 59.80 mm to 59.85 mm, the number of teeth of the inner rotor is nine, and the number of teeth of the outer rotor is ten. In this case, the same effects as the above embodiment can also be obtained.

Further, the above embodiment has been described about the construction employing a crankshaft directly-connected and driving method in which the inner rotor 10 is connected to a driving shaft directly connected to a crankshaft of an engine, and is driven by the rotation of the engine. However, the present invention is not limited thereto, and may be applied to, for example, an internal gear pump which employs a direct-current (DC) motor having a relatively small driving force as driving means, and which conveys fuel, such as light oil, having a relativity high viscosity. In other words, as previously described, it is possible to realize a remarkable decrease in the sliding resistance because the construction with no faucet can be realized without causing a problem such as the occurrence of

collision.

Moreover, as shown in FIG. 1, the above embodiment has been described about the construction in which the center O3 of the hole 31 of the casing 30 is disposed on a section of an extension line, which is obtained by connecting the axis O1 of the inner rotor 10 with the axis O2 of the outer rotor 20, located opposite to the axis O1 and the engaging position A with respect to the axis O2. However, the present invention is not limited thereto. For example, the center O3 may be disposed on a section of the extension line opposite to the axis O1 and the engaging position A with respect to the axis O2 so that the angle formed between a straight line, which is obtained by connecting the axis O1 with the center O3, located on a circumference having a radius eh of the inner rotor 10 centered on the axis O1, and the extension line is 0° to 30°.

In this case, the same effects as those in the above embodiment can also be obtained.

~~Industrial Usability~~

According to the present invention, it is possible to

provide an internal gear pump in which the sliding resistance of the internal gear pump can be reduced, and the occurrence of noise and a decrease in the pump efficiency can be suppressed to the minimum even in this construction.